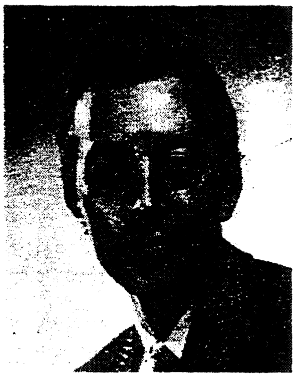
No. 1844

Heat and Moisture Flow Through Openings by Convection



W. G. BROWN

A. G. WILSON
Member ASHRAE

K. R. SOLVASON
Member ASHRAE

In calculating heating and cooling loads, it is often necessary to estimate the heat and moisture flow across openings in partitions that separate spaces at different conditions. These openings may be doorways or hatches used only infrequently, or they may provide more or less continuous connection between the two spaces for purposes of produce or cargo handling or for venting. A common example is the access for conveyor belts from cold storage warehouses to shipping areas. Heat and moisture flow can occur across such openings as a result of mass air flow due to natural convection forces. These forces will often predominate when no positive venting of the conditioned space is provided, or where there is no large imbalance of supply over exhaust air. This type of heat and moisture flow may be accentuated with multiple openings at different levels in the same or different walls of a conditioned space.

The 1961 ASHRAE Guide, in Chapter 24 on Infiltration and Ventilation, gives the elementary relationship for mass flow by natural convection through simple openings at two different levels in a building that results from differences in temperature between inside and outside air. This relationship is valid for openings with flow characteristics approximating those of a sharp-edged orifice, if the pressure difference across the openings is large relative to the pressure gradient over the height of the opening. This does not apply to single or multiple openings at one level, or to multiple openings at different levels, where the opening height is of the same order as the distance of the opening from the neutral level, that is, the level at which the pressures on either side of the partition containing the openings are equal.

Experimental data[1,2] applicable to this case and to natural convection through openings in horizontal partitions have recently become available, although not in a form suitable for engineering purposes. The need to estimate accurately the heat and moisture flow through such openings arises increasingly in the design of special purpose refrigerated and air-conditioned spaces. There is the associated problem of inhibiting such heat and moisture transfer. In this paper, relationships which are applicable to many of these problems are presented, along with charts to facilitate the calculations. The application of the charts to various types of problems is demonstrated.

EXPERIMENTAL WORK

In the experimental work referred to[1,2] measurements of the rate of heat transfer through openings were made with a large scale wall heat-flow measuring apparatus.[3] The apparatus (see also Fig. 1) consists of warm and cold boxes, each 8 ft square and 4 ft deep. The warm

W. G. Brown, A. G. Wilson and K. R. Solvason are with the Building Services Section, Division of Building Research, National Research Council of Canada. This paper, a contribution from the Division of Building Research of the National Research Council and published with the approval of the Director of the Division, has been prepared for presentation at the ASHRAE 70th Annual Meeting, to be held in Milwaukee, Wisc., June 24-26, 1963.

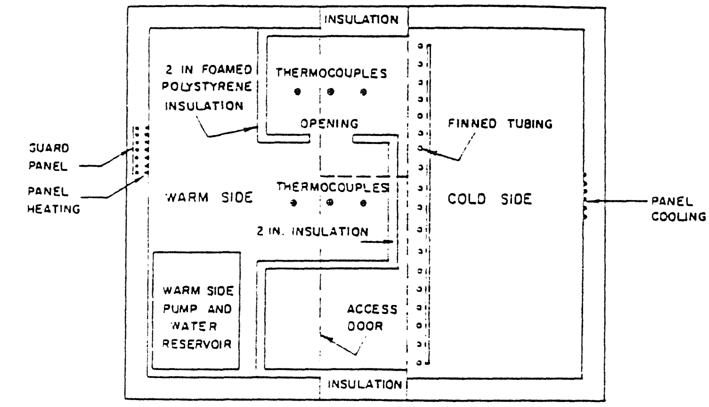


Fig. 1 Equipment arrangement for natural convection heat flow through an opening in a horizontal partition

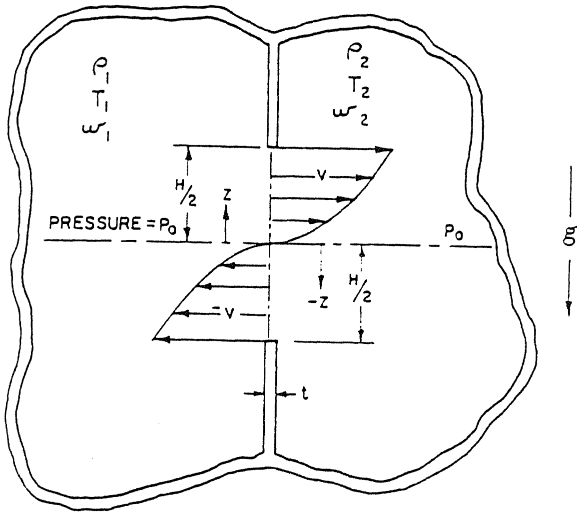


Fig. 2 Schematic representation of natural convection across an opening in a vertical partition

box is guarded and the heat input measured. The apparatus provides natural convection heat exchange on the warm side and either natural or forced convection on the cold side.

In the tests on vertical openings, heat transfer measurements were made on an 8 by 8-ft test wall of 2-in. thick foamed polystyrene, backed with ¼-in. plywood, into which were cut single openings of 6 by 6 in., 6 by 12 in., 9 by 9 in., and 12 by 12 in. Tests also were made with several 3 by 3-in. openings spaced horizontally and two 3 by 3-in. openings spaced vertically 15 in. apart on centers. The partition thickness in the vicinity of the openings also was varied to increase the number of combinations of thickness and opening height. Tests were carried out with air temperature differences across the wall ranging from 15 to 90 F, with the warm side at about 70 F.

The majority of measurements was made with natural convection heat exchange on both sides of the test wall; some tests were conducted with forced convection on the cold side, with air flowing horizontally and parallel to the wall surface at velocities of 100 and 200 fpm. The overall heat transfer coefficient of the test wall was first determined as a function of mean temperature for the three cold-side convection conditions. In subsequent tests with an opening in the test wall, the conduction heat transfer through the wall was subtracted from the total heat transfer to obtain the net heat transfer through the opening. This was then corrected for the heat transfer by radiation through the opening, assuming that the two enclosures behave as black bodies, to obtain the net heat transfer through the opening by air interchange.

The use of a test wall having a low thermal conductance increased the accuracy of the measurements, since a large portion of the total heat flow occurred through the openings. Furthermore, convection conditions would be expected to approximate those that would occur with density differences due to variations in the humidity ratio alone, so that the results also could be applied to moisture transfer.

In tests on horizontal openings, it was necessary to construct a special wall section incorporating a horizontal partition into which the openings could be cut. The test section of foamed polystyrene was built in the form of a 3-ft cubical box protruding from the wall, as shown in Fig. 1. In the majority of tests, the openings were cut in the upper partition, with the cold air above and warm air below; a few tests were made with an opening in the lower partition. Measurements were made with single openings of 6 by 6 in., 9 by 9 in., and 12 by 12 in., with the thickness of the partition varied from 1 to 8 in. One set of tests was conducted with a 12 by 12-in. opening and an 8-in. thick partition bevelled at a 45 deg angle to a 2-in. thickness. The temperature conditions and the method of obtaining the heat flow due to air interchange across the openings was the same as for the vertical position. All tests were carried out with natural convection in both warm and cold boxes.

RELATIONSHIPS FOR VERTICAL PARTITIONS

Fig. 2 represents a single opening of height H and width W in a vertical partition between two rooms at different air conditions, there being no other connection between the rooms. The densities of the air remote from the wall are maintained at ρ_1 and ρ_2, the temperatures at T_1 and T_2, and the humidity ratios at w_1 and w_2. Since there can be no net flow across the opening, the absolute pressure near the elevation of the center of the opening is equal on either side of the partition. In most practical cases, the difference in air density on either side of the partition is small compared with the mean density, and the level of equal pressure, or neutral level, can be assumed at the level of the center of the opening, with little error. A difference in pressure between the two rooms occurs at all points above and below the neutral level as a result of the difference in the density of the air on either side, the pressure difference varying directly with the distance from the neutral level. These pressure differences for single openings are relatively small but can produce significant air flow, as will be shown later.

The pressure difference be-

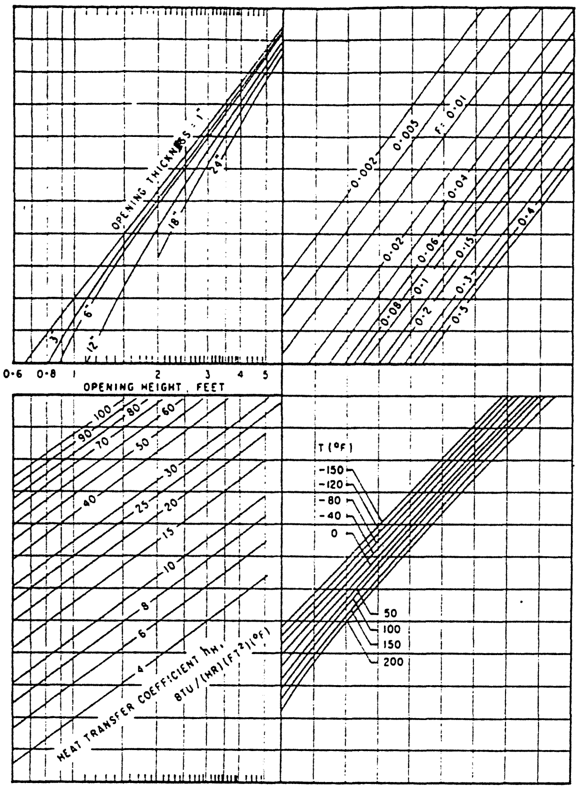


Fig. 3a Convection heat and moisture flow through a wall opening for $6 \times 10^{-12} < G < 6 \times 10^{-10}$

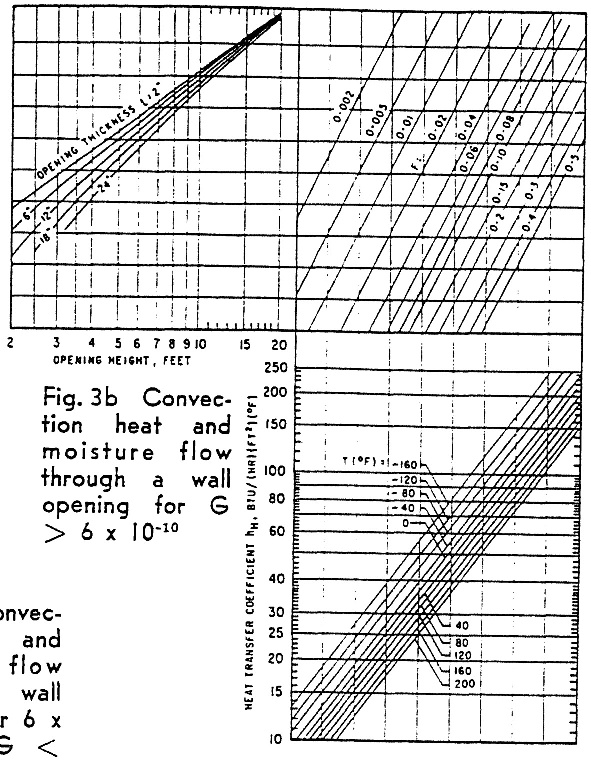


Fig. 3b Convection heat and moisture flow through a wall opening for $G > 6 \times 10^{-10}$

tween the two rooms at any level z below the neutral level is:

$$h_z = z\,(\rho_1 - \rho_2)\,(g/g_c) \times 12/62.4 \qquad (1)$$

where

h_z = pressure difference at level z below the neutral level, in. of water
z = distance below neutral level, ft
ρ_1, ρ_2 = density of air in the two rooms, lb/cu ft
g = gravitational constant, 32.2 ft/sq sec
g_c = dimensional constant, 32.2 lb mass ft/lb force/sq sec

The pressure difference at level z above the centerline is just the negative of Eq. (1). Air will thus flow in opposite directions above and below the neutral level with varying velocity. With this interchange of air, Q, between the two rooms, there is associated a net rate of heat transfer.

$$q = Q\,\bar{\rho}\,c_p\,(T_1 - T_2) \qquad (2)$$

where

q = net rate of heat transfer, Btu/hr
Q = effective rate of air interchange, cu ft/hr
$\bar{\rho}$ = average air density, lb/cu ft
c_p = specific heat of air at constant pressure, Btu/(lb) (F)
T_1, T_2 = air temperatures on either side of the partition, F

Similarly, there is a net rate of mass transfer of water vapor,

$$m = Q\,\bar{\rho}\,(w_1 - w_2) \qquad (3)$$

where

m = net rate of water vapor transfer, lb/hr
w_1, w_2 = mass fraction of water vapor on either side of the opening, lb water vapor/lb of mixture. For practical purposes, w_1 and w_2 usually can be taken as the humidity ratios, i.e.,/lb water vapor/lb dry air

A heat-transfer coefficient h_H and a mass transfer coefficient h_m for the opening can now be defined as:

$$h_H = q/WH\,(T_1 - T_2) \qquad (4)$$

where

h_H = heat transfer coefficient, Btu/(hr) (sq ft) (F)
W = opening width, ft
H = opening height, ft

and

$$h_m = m/WH\,(w_1 - w_2) \qquad (5)$$

where

h_m = mass transfer coefficient for water vapor, lb/(hr) (sq ft) (lb per lb)

Test results obtained for natural convection conditions on both sides of the test wall were expressed in terms of the appropriate dimensionless numbers in order to extend the application of the data. The following empirical relationship adequately expresses the data over the range of the test variables:

$$Nu/Pr = 1.16\left(1.0 - 0.6\frac{t}{H}\right) \times Gr^{0.432} - 215 \qquad (6)$$

where

t = partition thickness, ft
$Nu = h_H H/k$ (Nusselt number) (7)
k = thermal conductivity of air, Btu/(hr) (sq ft) (F per ft)
$Pr = \mu c_p/k$ (Prandtl number) (8)
(for air Pr has the constant value 0.71)
μ = absolute viscosity, lb/(hr) (ft)
$Gr = g\,\Delta\rho\,H^3/\nu^2\,\rho$ (Grashof number) (9)
$\Delta\rho$ = difference in density of air on either side of the opening, lb/cu ft
ν = kinematic viscosity, sq ft/hr

$$\frac{\Delta\rho}{\bar{\rho}} = F = \frac{T_1 - T_2}{T_{av}} + \frac{0.61\,(w_1 - w_2)}{(1 + 1.61\bar{w})\,(1 + \bar{w})} \qquad (10)$$

where the right-hand term is in differential form, T_{av} is the average absolute temperature and $\bar{w}$ is the average humidity ratio.

Eq. (6) has been used to construct Fig. 3a, from which values of the heat transfer coefficient can be obtained conveniently. Values of h_m, the mass transfer coefficient for water vapor, are related to h_H through Eqs. (2), (3), (4) and (5). Thus,

$$h_m = h_H/0.24 \qquad (11)$$

A useful relationship for the air interchange, Q, is obtained from Eqs. (2) and (4).

$$Q = h_H\,WH/\bar{\rho}\,c_p \qquad (12)$$

In using Fig. 3a, the limits of the test results should be kept in mind. The results are valid only for values of t/H between 0.19 and 0.75, and for values of the Grashof number between 10^6 and 10^8. In practical terms, this latter requirement can

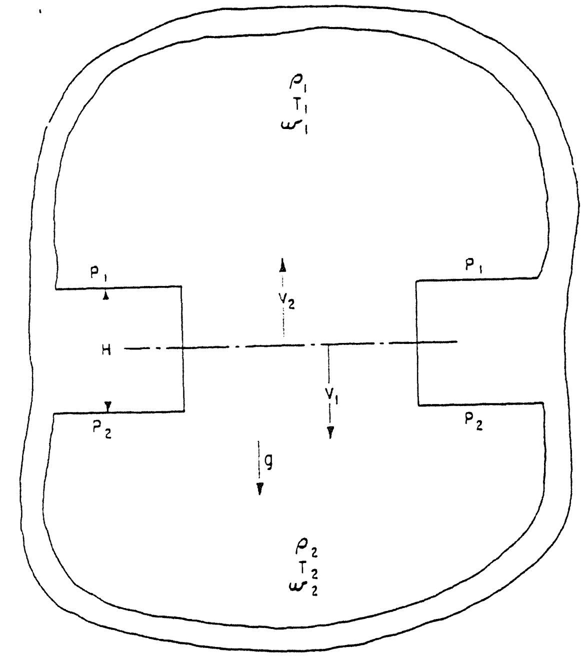


Fig. 4 Schematic representation of natural convection through an opening in a horizontal partition

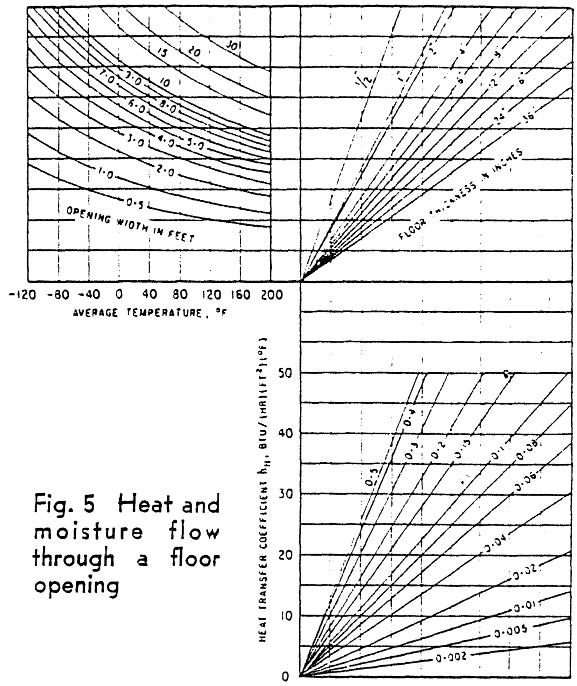


Fig. 5 Heat and moisture flow through a floor opening

be expressed approximately as:

$$G = \left\{ \frac{T_1 - T_2}{T_{av}} + \frac{0.61\ (w_1 - w_2)}{(1 + 1.61\ \overline{w})\ (1 + \overline{w})} \right\} \left(\frac{H}{T_{av}} \right)^3 = 6 \times 10^{-12} \text{ to } 6 \times 10^{-10}$$

Although this expresses the test range, it was found that at high values of Gr the test results approached the theoretical relationship $Nu/Pr = \frac{1}{3} Gr^{1/2}$ for small values of t/H. Consequently, the following modified theoretical equation was used to construct Fig. 3b, which should be suitably accurate for large values of G.

$$Nu/Pr = 0.343\ Gr^{1/2} (1 - 0.498\ t/H) \tag{13}$$

Examples of the use of Figs. 3a and 3b for both single and multiple openings and for the condition of displacement of the neutral level are given at the end of the paper.

The test apparatus contained means for creating a horizontal flow of air on the cold side of the partition at velocities of either 100 or 200 fpm. Since this flow was confined to a small region extending about 1 ft from the wall, the test results for this case would not be expected to have universal application. Results are given in Table I for comparative purposes; values for other configurations would be expected to differ. From Table I, it will be noted that whereas at 100 fpm the effect of natural convection is noticeable, at 200 fpm the heat transfer coefficient is practically constant for all opening sizes and is almost independent of natural convection forces. It is interesting to note that the heat transfer coefficient with natural convection was greater than that with a velocity of 100 fpm for all test conditions, and was greater than with a velocity of 200 fpm at large values of Grashof number.

RELATIONSHIPS FOR HORIZONTAL PARTITIONS

The flow condition for a square opening in a horizontal partition with the air above more dense than that below (as illustrated in Fig. 4) is inherently unstable and no steady distribution of flow and pressures can be assumed, as is possible for vertical openings. An average condition can be envisaged, however, in which air is interchanged across the openings, and heat and moisture transfer coefficients can be defined in terms of Eqs. (2), (3), (4) and (5). The results of the experimental work with the opening in the upper partition were adequately correlated by the following equation:

$$Nu/Pr = 0.0546\ Gr^{0.55} (L/H)^{1/3} \tag{14}$$

Table I Effect of Cold-Side Horizontal Air Flow on Heat Transfer Through Openings in Vertical Partitions

Opening		Air Temperature Difference, F			Mean Air Temperature, F			Heat Transfer Coefficient h_H (Btu/(hr)(F)(ft²))		
		Velocity, fpm			Velocity, fpm			Velocity, fpm		
Size, in.	t/H	0	100	200	0	100	200	0	100	200
6 × 6	0.38	30	32	31	57	56	56	14.9	7.9	22.7
6 × 6	0.38	48	51	49	48	46	47	19.7	8.2	22.1
6 × 6	0.38	66	70	68	38	36	36	22.6	10.4	22.1
6 × 6	0.38	84	88	86	29	27	27	26.2	13.2	22.2
6 × 12	0.38	29	31	30	57	56	56	13.8	8.5	20.1
6 × 12	0.38	47	50	50	47	46	45	18.6	9.7	20.3
6 × 12	0.38	70	75	73	35	37	33	24.1	12.9	20.3
9 × 9	0.25	21	22	21	62	61	61	15.8	7.7	22.1
9 × 9	0.25	29	—	30	57	—	56	19.5	—	22.0
9 × 9	0.25	46	—	49	47	—	45	25.5	—	20.1
12 × 12	0.19	26	20	19	57	61	61	23.9	8.3	19.4
12 × 12	0.19	33	30	29	53	56	56	28.8	10.0	19.4
12 × 12	0.19	37	38	38	50	51	51	31.4	10.9	18.9

where

L = opening width, ft
H = thickness of partition, ft

Nu, Pr and Gr are as previously defined, but with H as defined above. Eq. (13) has been used to construct Fig. 5, from which values of h_H, the heat transfer coefficient, can be obtained. The moisture transfer coefficient, h_m, is related to h_H by Eq. (11) as before. The use of Fig. 4 is clarified by examples at the end of the paper.

Eq. (13) and Fig. 4 are strictly valid only for the range of test conditions, which in this case was: H/L between 0.0825 to 0.66, and Grashof number (based on partition thickness H) between 3×10^4 and 4×10^7. In practical terms, the condition on the Grashof number is approximately:

$$G = \left\{\frac{T_1 - T_2}{T_{av}} + \frac{0.61\,(w_1 - w_2)}{(1 + 1.61\,\overline{w})\,(1 + \overline{w})}\right\}\left(\frac{H}{T_{av}}\right)^3 = 2 \times 10^{-13} \text{ to } 2 \times 10^{-10}$$

For field use, this range could probably be extended to about 10^{-8} without danger of essential error.

(Note $G \approx Gr \times 6 \times 10^{-18}$.)

The few tests carried out with the opening in the lower partition, in which the air below was more dense than that above, indicated a stratified air condition with pure conduction heat flow as would be predicted on thermal grounds.

RELATIONSHIPS FOR A LONG THIN SLOT OR NARROW TUBE

The previous results apply to relatively large openings in partitions in which the convection mechanism predominates over diffusion. The case of long openings with small cross sections can be handled theoretically, without the need for tests. The flow of heat or water vapor with diffusion alone can be calculated readily. Similarly, flow under conditions of total pressure differences across the opening can be determined from known relationships for pipes or tubes. There remains the case of heat and mass flow due to combined diffusion and convection. Assuming entirely viscous flow in the opening and neglecting entrance and exit effects, the basic flow equation for mass transfer can be written approximately as:

$$V_{av}\frac{dw}{dx} = D\frac{d^2 w}{d x^2} \qquad (15)$$

where

V_{av} is the average velocity
x is the distance along the slot
D is the diffusion coefficient
w is the humidity ratio

Integration is a simple matter. With air entering the slot with humidity ratio w_1 and leaving at w_2, the total rate of water vapor flow through the slot of cross-sectional area A is, by integration,

$$m = \overline{\rho}\,A V_{av}\left[w_1 - \frac{(w_2 - w_1)}{e^{(V_{av} t/D)} - 1}\right] \qquad (16)$$

In practice, the velocity, V_{av}, arises as a result of pressurizing one side of the slot either by artificial means, or as a result of buoyancy effects acting on multiple slots or openings in a partition. An example is given in the next section.

Generally, the flow through a slot is so small that the flow of heat in it cannot be represented precisely by a basic equation of the form of Eq. (15). For estimation purposes, however, Eqs. (15) and (16) can be used for heat transfer simply by substituting the thermal diffusivity α for D, and T (absolute temperature) for w, and multiplying Eq. (16) by c_p.

APPLICATION OF RESULTS

Example No. 1

A single 4 by 4-ft opening in an 8-in. thick wall between a cold storage room at 0 F and 90% rh and a warehouse at 60 F and 40% rh. From psychrometric charts, the humidity ratio, w_2, in the cold room is 0.0006 lb/lb and in the warehouse w_1=0.0044. The factor,

$$F = \left\{\frac{T_1 - T_2}{T_{av}} + \frac{0.61\,(w_1 - w_2)}{(1 + 1.61\,\overline{w})\,(1 + \overline{w})}\right\}, \text{ has the}$$

value of 0.125 and G has the value of 6.8×10^{-8}. Entering Fig. 3b at the top left with H = 4 ft and t = 8 in., a horizontal line is projected to the right, meeting the line F = 0.125. A vertical line is then projected to the average temperature of 30 F, and from there a horizontal line to the left gives a value for h_H = 88 Btu/(hr)(sq ft)(F) [h_m = 88/0.24 lb/(hr)(sq ft)(lb/lb)].

The heat flow by convection across the opening is:

$$h_H WH\,(T_2 - T_1) = 88 \times 16 \times 60 = 85{,}000 \text{ Btu/hr}$$

The moisture flow across the opening is:

$$h_m WH\,(w_2 - w_1) = \frac{88}{0.24} \times 16 \times 0.0038 = 22 \text{ lb/hr}$$

Example No. 2

A 4 by 4-ft opening as above but in an 8-in. thick floor. In Fig. 5, enter at the top left with the average air temperature of 30 F and extend vertically to the opening width of 4 ft. From this point, extend horizontally to the right to meet the line of floor thickness, H = 8 in., and drop a vertical line to the value of F = 0.125. Extending then to the left gives h_H = 26 Btu/(hr)(sq ft)(F). In this case, the heat flow by convection is 25,000 Btu/hr and the moisture flow is 6.6 lb/hr.

Example No. 3

The conditions on the two sides of a partition are 70 F, 90% rh and 74 F and 7% rh, corresponding to w_1 = 0.0142 and w_2 = 0.0016, respectively. The factor F has the value $\dfrac{70 - 74}{532} \times \dfrac{0.61(0.0126)}{(1.013)(1.008)}$ = 0. In this case, there is no convection across an opening and heat and moisture flow have only nominal values, since they occur by conduction and diffusion, respectively.

Example No. 4

An open door 2.5 ft wide by 6.5 ft high in a 12-in. wall separates spaces at 32 and 70 F; the humidity ratio on both sides of the door is the same. It is required that no 70 F air enter and contaminate the 32 F space. In this case, the position of the neutral level must be displaced from an elevation of one-half the door height to the full height of the door. The pressurization required to effect this [see Eq. (1)] is $h_z = \dfrac{(0.0806 - 0.0748)}{62.4}\,12 \times \dfrac{6.5}{2}$ = 0.0037 in. of water. The flow rate is related to the heat transfer coefficient (which can be obtained from Fig. 3b) by Eq. (12), i.e.,

$$Q = h_H \, WH/\bar{\rho} \, c_p, \text{ cu ft/hr}$$

To find h_H from Fig. 3b, it must be recognized that H is twice the distance from the neutral level to the upper or lower boundary of the openings; in this instance, the neutral level is at the top of the doorway, so that H = 13 ft. With $F = \frac{38}{511} = 0.074$, $h_H = 124$ Btu/(hr)(sq ft)(F). Hence $Q = \frac{124 \times 2.5 \times 13}{0.078 \times 0.24} = 215{,}000$ cu ft/hr.

Example No. 5

Multiple openings in a wall. A 2 by 2-ft opening and a 3 by 3-ft opening are arranged in a 6-in. wall in such a way that the bottom edge of the smaller opening is 1 ft above the top edge of the larger opening. The humidity ratios on both sides of the wall are equal but the two air temperatures are 20 and 50 F (F = 0.06). The problem is to find the location of the neutral level by trial and error use of Fig. 3.

First trial — Assume that the neutral level lies 0.25 ft below the upper edge of the 3-ft opening. In this case, there are four values for H corresponding to the distance of the neutral level from the top and bottom of each opening. The values of H are 6.5, 2.5, 0.5 and 5.5 ft, for which the values of h_H are 80, 47, 18 and 73, respectively. (Note that $G = 6.2 \times 10^{-11}$ for H = 0.5 ft and h_H is obtained from Fig. 3a in this case.)

From Eq. (12), the air flow in one direction is 44,400 cu ft/hr, and in the other direction the flow is 60,500 cu ft/hr. Since these two figures are not equal, the neutral level is obviously lower than estimated. Assuming values of H = 7.0, 3.0, 1.0 and 5.0 ft, for which the values of h_H are 83, 43, 22 and 69, the air flow in one direction is 51,800 cu ft/hr and in the other direction the flow is 52,000 cu ft/hr, corresponding to a heat flow of 30,000 Btu/hr.

Example No. 6

Two horizontal service pipes, ½ in. inside dia and 12 in. long, are separated by a vertical distance of 1.0 ft in a wall. With the temperature everywhere constant at 70 F and with $w_1 = 0.014$ and $w_2 = 0.002$, what is the net moisture flow rate between the two sides of the wall?

The pressure producing flow is [from Eq. (1)]:

$$h_s = 0.5 \times 0.61 \; (0.012)/(1.013)(1.008) = 0.0036 \text{ ft of air}$$

From Darcy's equation of laminar flow,

$$h_s = \frac{64\,\nu}{V\,d} \cdot \frac{t}{d} \cdot \frac{V^2}{2\,g}$$

Where d is pipe dia and t its length, V is found to be 140 ft/hr. The diffusion coefficient D has a value of about 1.0 sq ft/hr. The exponent on e in Eq. (16) has the value 140, hence the second term can be disregarded. In other words, diffusion plays a negligible role in this specific problem. The net mass flow from Eqs. (16) or (3) is 0.0002 lb/hr.

CONCLUSION

Equations and charts have been presented, with which it is convenient to calculate the heat and moisture transfer due to natural convection through openings in vertical partitions separating spaces at different air conditions. These apply specifically to the case of single and multiple openings having a thickness which is small relative to height, and where the opening height is of the same order as the distance of the opening from the neutral axis, for which experimental data have only recently become available. The heat and moisture transfer coefficients depend on the Grashof number, and to some extent on the ratio of opening height to thickness. Example calculations are given which demonstrate that heat and moisture transfer through such openings can be relatively large under many practical conditions.

A method of determining the total flow required to prevent any air interchange across the openings also is described. Some test results are given which indicate the effect of forced convection across one side of the wall containing the opening. Such forced convection may either increase or decrease the net interchange across the opening.

A chart and equations also are given for calculating heat and moisture flow due to natural convection across an opening in a horizontal partition, for the case where the higher density air is above the opening. The heat and moisture transfer coefficients again depend on the Grashof number, and increase with increasing ratio of partition thickness to minimum opening width. Example calculations indicate that flow across such openings can be significant.

An equation is given for the flow of moisture through small openings, in which the length is large in relation to cross section and where both convection and diffusion mechanisms are present.

REFERENCES

1. Brown, W. G., and K. R. Solvason, Natural Convection through Rectangular Openings in Partitions. Part I—Vertical Partitions, Int. J. Heat and Mass Transfer, Vol. 5, September 1962, pp. 859-868.

2. Brown, W. G., Natural Convection through Rectangular Openings in Partitions. Part II—Horizontal Partitions (op. cit.).

3. Solvason, K. R., Large Scale Wall Heat Flow Measuring Apparatus, ASHRAE Transactions, Vol. 65, pp. 541-550 (1959).

DISCUSSION

C. W. PHILLIPS, Washington, D. C.: In refrigerated vehicles used for frozen food delivery, the problem of heat exchange through an open door may transcend the body loss to the point where body loss is a relatively unimportant item in the total. However, the opening of a door in a container like this is not a steady state operation, from the standpoint of either time or of temperature difference. Can you suggest an application of your data to the transient state where the temperature inside the body will begin to adjust to the outside temperature as the door opens? That is, the non-steady state transfer of moisture and heat through a vertical opening.

AUTHOR BROWN: We, ourselves, haven't done any work on this. I believe Professor Schmidt in Germany has done a small amount of work of this type, but it was done, I believe, with CO_2 in air.

We could, with our apparatus, do transient tests. I am not certain, myself, that is, I have not looked into the matter, whether or not we could, with the smaller openings that we had, extrapolate these results to more normal sizes. Other than this, I don't have any immediate information. Since air adjusts rapidly to changes in conditions, however, it may be possible for calculation purposes to assume that the instantaneous rate of heat or mass transfer in the transient case is the same as that of the steady-state.

P. R. ACHENBACH, Washington, D. C.: Considering the dimensions and size relationships indicated in Fig. 1, do you believe that the proximity of the enclosing walls on the cold and warm sides of the horizontal opening influenced the magnitude of the air exchange and heat exchange? To what extent did the finned tubing on the cold side interfere with the exchange of air and heat between the cold side of the apparatus and the cavity above the horizontal opening? Would you expect that the air exchange between the cold side of the apparatus and the cavity above the horizontal opening created velocity patterns that would influence the air and heat flow through the horizontal opening?

AUTHOR BROWN: Your first remarks are quite correct about the boundaries, with reference to Fig. 1 in particular. These openings are in a confined space and the local boundaries, the nearness of the ceiling of the test apparatus and all such factors would have a definite influence. We did not go further with our tests but feel that these effects are relatively small. The same situation exists for the vertical partition, i.e., the nearness of the thin coil. However, these appear to be the first test results of this kind and we would support them for use in an approximate manner.

As I pointed out, any additional geometrical differences between our conditions and those in the field would influence the results and I don't claim high accuracy for this work, but I feel it to be quite appropriate for any practical application. In the case of recirculation, of air on the cold side, I should present again the test conditions that we had; either velocity of 100 fpm or 200 fpm over the cold side in the horizontal direction. At the same time, there was a temperature difference between the cold and the warm side. Generally speaking, with 100 fpm air velocity, the coefficient of heat transfer across the opening was decreased, whereas, with the higher flow of 200 fpm it was increased above that for the natural convection alone. I am not able to explain the results.

MR. PHILLIPS: I think that it is important to differentiate between spaces at two different temperatures that might have leaks which permit air to be flowing in one direction only through such an opening. With a truck body, for example, the door so completely dominates leakage that you are forced to have air flow in both directions to satisfy the net balance of air, but in a cold storage warehouse, this is not apt to be the case. You are apt to have leaks other than those that occur when you open the door. The air flow through the door can be all in one direction and the flow will be quite different from the conditions which you described in these sealed box tests.

AUTHOR BROWN: Yes, this is quite correct and, as I mentioned during presentation of the paper, it is possible to superimpose the two solutions. That is, if you have information on the net through flow or the net pressurization of your truck, you can calculate the combined effect of both natural and forced convection. We have one or two examples of that type in the latter part of the paper.

In closing, I want to apologize for a mistake in the preprint. Factor F is simply the difference in density divided by the mean density. Would you neglect the other expression until we have the final paper?